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PATENT SPECIFICATION

DRAWINGS ATTACHED

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COMPLETE SPECIFICATION

Variable Ratio Friction Gears

WE, THE ENGLISH ELECTRIC COMPANY LIMITED, of English Electric House, Strand, London, W.C.2, a British Company, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:

This invention is concerned with variable-ratio frictional drive gears of the kind comprising basically two axially spaced torus discs between which there is a set of circumferentially spaced drive rollers in frictional rolling contact with toroidal surfaces on the discs, each roller being rotatably mounted in a roller carriage which can tilt about an axis at right angles to the axis of rotation of the roller so as to vary the distances from the gear axis at which the roller engages respectively the two discs, thus varying the drive ratio of the gear. The angle of tilt of the roller carriages, as it controls the drive ratio of the gear, is called the "ratio angle".

One way of changing the ratio angle is to tilt the roller carriages by means of a positive mechanical linkage. This invention is however concerned with an alternative arrangement in which this control is achieved indirectly by bodily moving the roller carriages in substantially tangential directions with respect to the gear axis, and by allowing the rollers then to steer themselves towards a different ratio angle; gears of this general construction will be referred to as "gears with tangentially controlled roller carriages". This invention is more specifically concerned with hydraulic control of gears of this general construction.

This invention is particularly though not exclusively concerned with gears in which the plane of each roller, normal to the axis of rotation of the roller and passing through the points of contact of the roller with the two opposed torus discs, contains the axis about which the roller tilts, being tangential to the torus centre circle (i.e. the locus of the centre of the circle revolved to generate the torus), as distinct from gears in which the same plane for each roller is closer to the main axis of rotation of the gear. The first arrangement, to which this invention is particularly applicable, has its rollers lying diametrically across the torus circle (and may accordingly be referred to as a "diametrical-roller gear") as opposed to the second arrangement, which has chordal rollers of smaller diameter than the torus circle.

A gear according to this invention comprises two axially spaced torus discs between which there is a set of circumferentially spaced drive rollers in frictional rolling contact with toroidal surfaces on the discs, each roller being rotatably mounted in a tangentially controlled roller carriage having end portions lying on a roller tilt axis at right angles to the axis of rotation to the roller, the end portions of each roller carriage being slidably and rotatably supported by the arms of a spider member whereby the roller carriages can be moved tangentially in order to control indirectly the ratio angle of the rollers, one end portion of each roller carriage having a piston member which is slidably mounted in a cylinder in the spider arm supporting that end portion, the cylinder being in communication with a passageway through which a controlled fluid pressure can be supplied to control the tangential position of the roller, the tilt axis of each roller being inclined by a camber angle to a plane normal to the gear axis,

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whereby a given amount of tangential shift of all the rollers results in a ratio angle change directly related to the amount of tangential shift. The camber angle is preferably between 6° and about 15°, a preferred value being approximately 12°.

This invention will be explained with reference to the accompanying drawings which show a preferred example of a diametrical-roller gear according to this invention. In these drawings:—

Fig. 1 shows the gear in longitudinal cross-section through the axis of the gear; and

Fig. 2 is a partly sectioned side view showing the arrangement of the rollers and roller carriages in their mountings.

Referring now to the drawings, the gear shown generally at 10 provides a variable ratio coupling between an input shaft 12 and an output shaft 11 and is supported in position by the shafts 11 and 12 and by a ball race assembly generally shown at 13.

Carried by the shaft 11 are two torus discs 14 and 15, of which the disc 14 is solidly keyed to the shaft 11, while the disc 15 is splined on the shaft 11 and is axially movable along the shaft.

Around the shaft 11 there is a sleeve member 16 which is part of a composite spider member shown generally at 17. Rotatably mounted on the outside of the sleeve member 16 is a third torus disc 20, which is axially movable along the sleeve member. Movably splined to the disc 20 is a drum 21 which extends over the torus disc 15 and is connected to the shaft 12.

Formed on both faces of the torus disc 20 and on the inner faces of the torus discs 14 and 15 are toroidal surfaces between which two sets of rollers (each set consisting of three rollers) are interposed in a back-to-back arrangement. The rollers of the set interposed between the torus discs 14 and 20 are denoted by the reference numeral 22 and those of the set between the torus discs 15 and 20 are denoted by the reference numeral 23.

Each roller is mounted in ball races 24 in a roller carriage 25, which is supported as is described below by parts 35 and 37 of arms which form part of the composite spider member 17. The composite spider member is in two parts; one part, which may be termed the "main spider", consists of arms 26 supporting the rollers 22 and including the integral sleeve 16, while the other part supports the rollers 23 and is in the form of an auxiliary spider comprising arms 27 and is mounted on the end of the sleeve 16 with means (not shown) preventing rotation of the auxiliary spider while permitting the axial movement necessary for allowing the rollers 23 to take up the appropriate positions in relation to the torus discs 15 and 20. The arms 26 of the main spider serve to support the composite spider member 17 by providing means (not shown) for securing it to the casing of the gear.

The roller carriages of the two sets of intermediate rollers are identical, each being shown in Fig. 2 as having arms 30 and 31 formed with part-spherical ends 33 and 32, the arm which is the leading of the two as regards the normal direction of rotation of the input shaft 11, (shown arrowed in the drawing), being denoted by 30, and the arm which is the lagging of the two being denoted by 31.

Each arm end 32 is slidably accommodated in a cylindrical bore 34 formed in the part 35 of the arm of the spider member 17. Each arm end 33 fits in a socket formed in a piston member 36 slidable in a cylindrical bore in the part 37 of the arm of the spider member 17.

The ball ends 32 and 33 are each arranged to be slightly offset from the plane parallel to the torus discs and which includes the roller centres, and each roller carrier is therefore slightly angled to this plane by a camber angle A (sometimes known as a "castor angle"), the purpose of which is later to be explained. In other words, the tilt axis of each roller is inclined by the angle A to a plane normal to the axis of the gear.

Between each piston member 36 and the back of its associated cylindrical hole is a cylinder space 40 which communicates through an oil passage 41 formed in the respective arm of the spider member with a common hydraulic control circuit which is arranged as is now to be described.

The common hydraulic control circuit is supplied from an oil reservoir (not shown) through an oil pipe 42 and comprises a pump 43 which supplies a constant oil pressure via a second oil pipe 44 to a hydraulic governor generally shown at 45.

The hydraulic governor 45 is responsive automatically to one or more of the parameters of the gear and associated apparatus, for example, speed or torque, or it may be manually controlled or subject to both automatic and manual control.

The governor effects control of the gear by variation of the oil pressure supplied to the gear through a third oil pipe 46, the latter being connected to oil passages 41 in the arms 26 by a further oil passage 47.

The oil passages 41 formed in the arms 26 and 27 communicate with one another through an oil chamber 50 formed on the inside of the sleeve member 16 as shown, and thereby oil of variable pressure may be supplied to the space 40 behind each piston member 36, and control of the gear ratio may be effected as will be described in detail later.

Communicating with the oil chamber 50 of the common hydraulic control circuit

through oil passages 51 and 52 formed in the driving shaft 11 is a space 53 between the back of the torus disc 15 and a cylinder member 54 secured to the shaft.

The cylinder member 54 has a flange portion 55 which fits closely over the edge of the torus disc 15 and so forms with the disc a piston and cylinder assembly, it being remembered that the disc 15 can move axially on the shaft 11.

A sealing ring 56 carried in a groove in the disc assures an oil-tight sliding fit between disc 15 and flange 55.

An increase in the oil pressure supplied through the pipe 46 causes each roller carriage to be pushed tangentially away from its cylinder 40, and such tangential shifting causes the rollers to steer themselves into a new ratio angle, the magnitude of the ratio angle change for any given amount of tangential shift being determined by the camber angle A: in the example shown, the angle A is 6°, but it may in practice set at any value between 0° and about 15°, a preferred value being approximately 12°.

In a similar way a decrease in the oil pressure causes each roller to tilt in the opposite direction into a new ratio angle by allowing the roller and roller carriage to be dragged tangentially under the action of its torque reaction force, such movement occurring until the torque reaction force is again balanced by the force due to the oil pressure applied to the piston members 36.

Torque equalisation between all six rollers of the gear is automatically effected by the hydraulic circuit inter-connecting the spaced 40 behind the piston members 36. Any roller caused for any reason to transmit a torque which differs from that corresponding to the pressure of the oil in this hydraulic circuit moves tangentially until it has tilted to a new ratio angle position at which equilibrium is restored.

During constant speed, constant torque operation of a load driven the shaft 11, the pump 43 and hydraulic governor 45 serve to provide a constant pressure to the piston members 36 to maintain the rollers in their desired positions.

It will be appreciated that the torque reaction thrusts of the rollers (that is to say, the torque tending to rotate them bodily around the axis of the gear) are referred to piston members where they are opposed by forces due to the pressure of oil in the common hydraulic circuit, and a particular oil pressure in the common hydraulic circuit therefore corresponds to a particular value of torque reaction on the rollers. The oil in the common hydraulic circuit is also supplied to the back of the torus disc 15. Accordingly any variation in the oil pressure, corresponding to a variation of torque reaction, results in a proportional change in the axial force provided.

By thus providing the common hydraulic circuit the gear shown in the drawing is provided with an axial force which is always appropriately related to the torque reaction on the rollers. The magnitude of this axial force is dictated by the piston areas of the torus disc 15 and of each piston member 36, and these areas are therefore chosen so that the axial force will be just sufficient for creating the reaction needed for transmitting the torque required at various loads and speeds. This should take account of the effect of lubricating oil on the coefficient of friction between the rollers and torus discs if the rollers are not allowed dry contact with the discs.

The piston areas of the torus disc 15 and the piston members 36 are chosen so that the axial force provided at the 1:1 ratio position is greater than that required to provide limiting friction only by the small extra force needed to give the safety factor considered necessary. The safety factor must be provided to make allowance for such variable factors as gear wear, component tolerance and torsional vibration.

When the rollers are required during operation of the gear tilt away from the 1:1 ratio position then, owing to geometrical considerations, the component of axial force acting along the roller; (i.e. the normal to the contact point of roller and torus disc) will be increased by a factor of $1/\cos \Theta$, where Θ is the angle through which each roller tilts from its 1:1 position. The direct result of this geometrical "cosine effect" is to cause a somewhat greater contact load between roller and torus disc at large angularity positions of the roller than that necessary, but the excess is insufficiently large for the extra gear wear involved to be serious.

Power transmission must be from the shaft 12 to the shaft 11, but the gear could be modified to allow the shaft 11 to drive the shaft 12 (in reverse directions to those shown) by setting the camber angles in the opposite senses. It should be noted that no control will be affected by the hydraulic governor in the event of overhauling by the load. This is because the torque reaction thrust exerted through its associated roller carriage by a roller on its piston member 36 must be positive. A negative torque reaction thrust, such as would occur during overhauling, would cause each roller to shift tangentially until prevented from further movement by abutment of the associated end 33 against the back of its co-operating hole 34, and no control is therefore possible under these conditions.

Gears according to this invention are particularly useful in constant speed drives for aircraft alternators. When used for this

purpose, the gears can be as shown in the drawings in having only one piston 36 for each roller carriage, and overhauling by the alternator can be prevented by the use of a sprag clutch interposed between the gear and the alternator. The rollers are preferably lubricated by oil injected through a central bore in each end portion 32 of the roller carriage, via the bore 34; this makes for a particularly neat and convenient arrangement.

By way of explanation it may be noted that the gear shown in the drawings can only operate in the manner described, that is to say with the shaft 12 serving as the input and rotating in the direction shown, because stable operation cannot be achieved in any other mode of operation. The first criterion is that the input member must rotate in the direction in which the driving torus disc (i.e. the disc 20) tends to drag each roller carriage towards its piston support 37 to enable the tangential position of the carriage to be controlled by fluid pressure in the cylinder space 40. (Clearly such control cannot be achieved by rotation of the disc 20 in the opposite direction because the fluid in the cylinder space 40 cannot exert a negative control pressure). The second criterion is that the camber angle must be in the sense shown, that is to say such that each roller carriage tilt axis is inclined away from the input disc (i.e. the disc 20) in the direction of movement of the disc. This criterion arises out of the fact that stable operation at any given ratio angle occurs when the axis of rotation of each roller passes through the common centre axis of the torus discs; unless the camber angle is in the sense just referred to, tangential displacement of a roller carriage (by virtue of an increase or decrease in the load on the gear or in the fluid pressure in the cylinder space 40) will result in the torus discs producing a steering force on the roller which will tilt the roller carriage in the direction opposite to that which is required to move the roller axis back to intersect the centre axis, so that the roller will be moved away from, instead of towards, its new stable position.

Whilst the invention has been described in relation to hydraulic operation of the various pistons, alternatively pneumatic operation could be employed.

The gear shown in the accompanying drawings embodies inventions which are the subject respectively of our pending patent applications Nos. 9962/65 and 35635/67. (Serial Nos. 1146321 and 1146324).

WHAT WE CLAIM IS:—

1. A variable-ratio frictional drive gear comprising two axially spaced torus discs between which there is a set of circumferentially spaced drive rollers in frictional rolling contact with toroidal surfaces on the discs, each roller being rotatably mounted in a tangentially controlled roller carriage having end portions lying on a roller tilt axis at right angles to the axis of rotation to the roller, the end portions of each roller carriage being slidably and rotatably supported by the arms of a spider member whereby the roller carriages can be moved tangentially in order to control indirectly the ratio angle of the rollers, one end portion of each roller carriage having a piston member which is slidably mounted in a cylinder in the spider arm supporting that end portion, the cylinder being in communication with a passageway through which a controlled fluid pressure can be supplied to control the tangential position of the roller, the tilt axis of each roller being inclined by a camber angle to a plane normal to the gear axis, whereby a given amount of tangential shift of all the rollers results in a ratio angle change directly related to the amount of tangential shift.

2. A gear according to claim 1, for which the axis about which each roller tilts to change the ratio angle coincides with the axis of the cylindrical bore in which the piston member slides.

3. A gear according to claim 1 in which end portions of each roller carriage terminate in part-spherical ends lying with their centres on the tilt axis of the roller carriage, the piston for each roller carriage having a part-spherical recess in which the corresponding end engages.

4. A gear according to any one of claims 1 to 3 in which the camber angle is between 6° and 15°.

5. A gear according to claim 4 in which the camber angle is approximately 12°.

6. A diametrical-roller gear according to any one of claims 1 to 5.

7. An aircraft alternator constant speed drive mechanism incorporating a gear according to any one of claims 1 to 6, the gear being used to transmit a variable speed drive by which speed variations of the aircraft engine are counteracted so as to produce a constant speed drive to the alternator.

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Chartered Patent Agent.

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This drawing is a reproduction of the Original on a reduced scale.

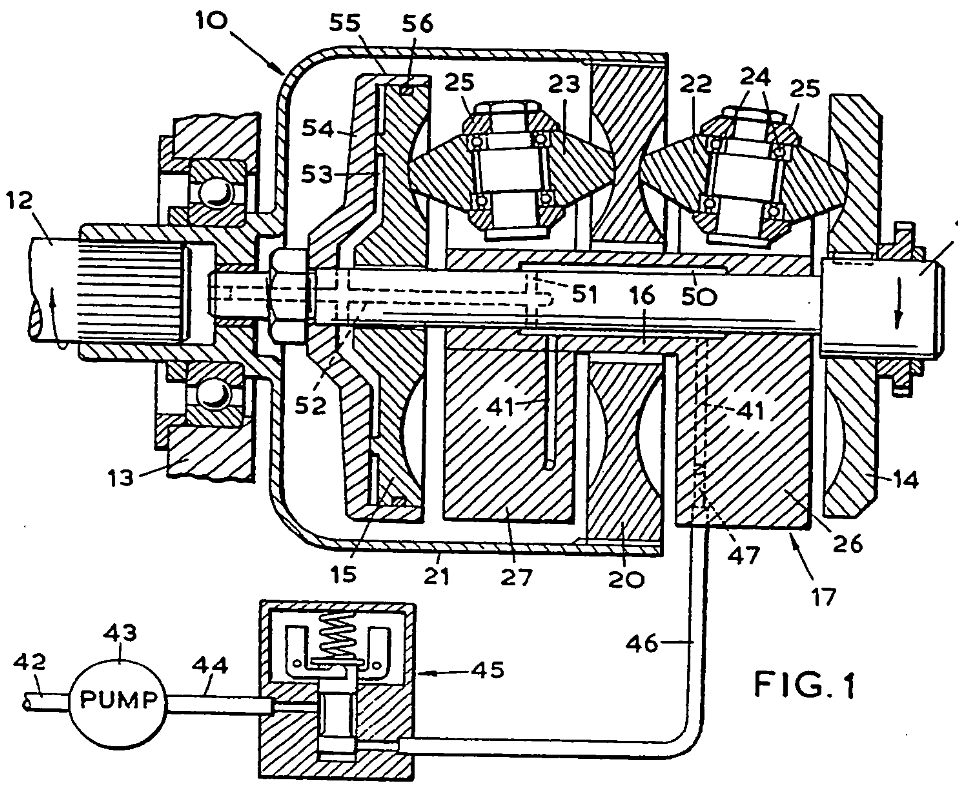
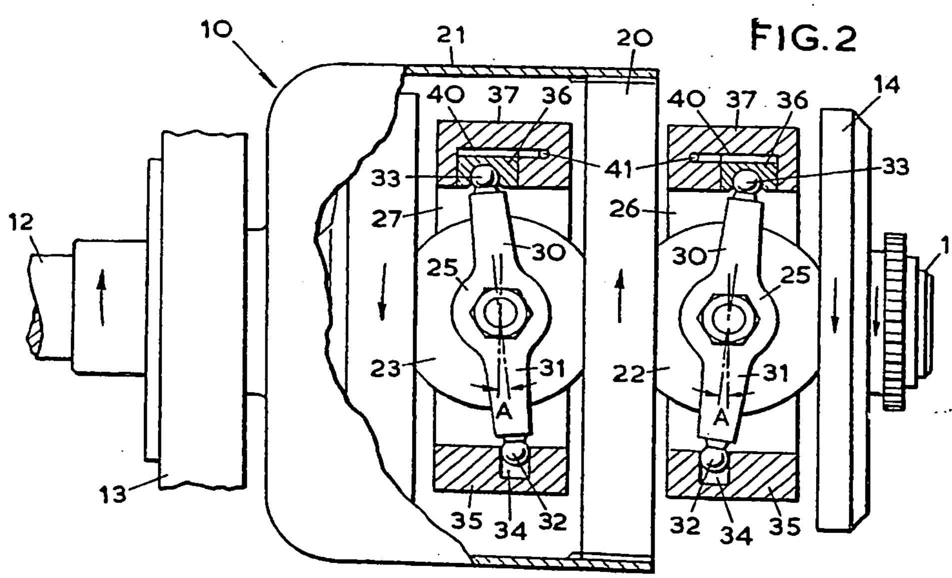


FIG. 1

FIG. 2



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